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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

This Issue

BALL AND ROLLER BEARING LUBRICATION

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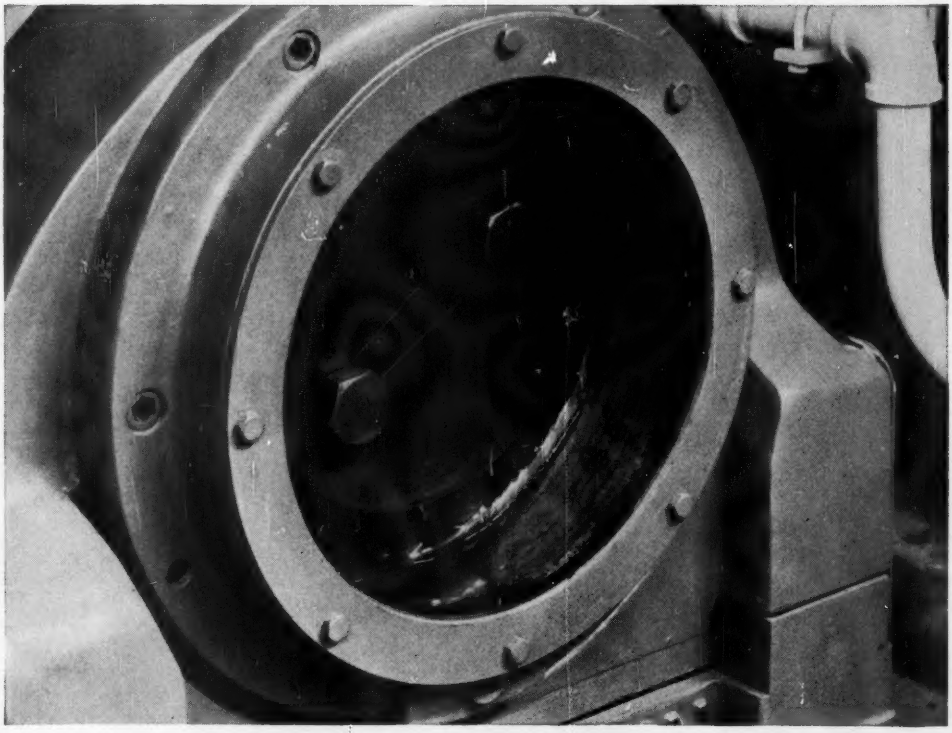

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LUBRICATION

A TECHNICAL PUBLICATION DEVOTED TO THE SELECTION AND USE OF LUBRICANTS

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Ball and Roller Bearing Lubrication

THE turn of the twentieth century was achieved smoothly in the engineering sense, as the common usage of rolling-element bearings dates back to about 1900[1]. Since that time these essentially simple antifriction devices have been developed to a high degree of performance in a multitude of specialized designs.

Children continue to roll along on ball-bearing skates, but the layman has little conception of the diversified applications of antifriction bearings in the world today. Such would be brought home to him if he could see with X-ray eyes into his household appliances, car and lawnmower. Electric motors turn easier, automobiles roll easier, and hand mowers push easier because of antifriction bearings at critical points. And in commerce, our industrial plants, trains, buses, trucks and airplanes use rolling-element bearings by the millions.

Modern ball and roller bearings are part of a team: the other part is the lubricant. The bearings could not do their jobs if it were not for the highly developed greases and oils that lubricate and protect them in service. This issue of magazine LUBRICATION describes advances in ball and roller bearing (BRB) lubrication during recent years.

WHY ANTIFRICTION BEARINGS NEED LUBRICATION

An ideal antifriction bearing would be one in which only pure rolling motion was encountered. However, this does not work out in practice. In service and under load, uncaged balls or rollers rub or slide somewhat against themselves. When a separator or cage is present, the rolling elements slide against this, and the cage itself rubs against any guiding flange surfaces.

The rolling elements may also show relative slip against the raceways. For instance, when a ball under load rolls in a curved groove, such as a curved bearing raceway, pure rolling is obtained only along two lines in the contact area; other parts of the ball slide or spin along areas of the groove to some extent. This is because the *effective* radius of the ball is smaller at points distant from the bottom of the race groove. Because of this sliding, lubrication is needed to minimize wear and friction. Loading intensifies this condition.

NATURE OF GREASE LUBRICATION

The great majority of rolling-element bearings are now lubricated with grease[1]. Being a solid when not under shear, grease forms an effective collar at bearing edges, sealing against dirt and moisture, and minimizing leakage as compared to the use of oil. Simpler housing designs are possible, and piping may be greatly reduced when grease is the lubricant.

Why doesn't grease "gum up the works" when it gets on the rolling and rubbing surfaces? Because

[1]See bibliography to identify this and subsequent numbered references.

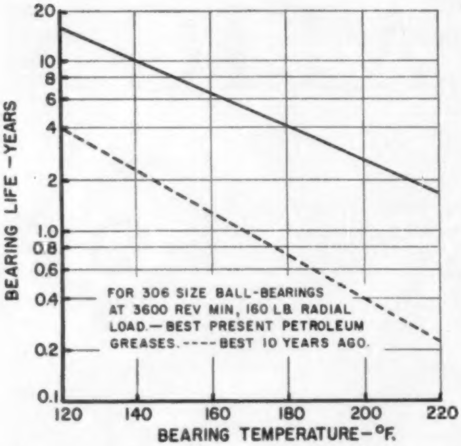


Figure 1 — Variation in Grease Life With Temperature (reprinted from 1957 article "Recent Advances in Grease Lubrication of Ball-Bearings" by E. R. Booser[5]).

grease becomes a fluid when intensively sheared or worked; therefore, the bearing is, in effect, lubricated with an oily film at points of dynamic contact. On the other hand properly compounded and uncontaminated greases are "thixotropic," which may be defined as softening under shear but returning to a solid when the shearing force is relieved.* It is easy to see from such a combination of properties why greases are widely used lubricants. They flow at the contact surfaces, but seal at the edges of a bearing.

It has been indicated for some time that certain prepacked, sealed or shielded rolling-element bearings, or those in which a controlled excess of grease is placed in the housing, are essentially lubricated by an oily phase which "bleeds" from the static grease to the moving surfaces[2, 3]. The bearing needs more oil from the grease as the following increase: bearing size, speed, load or temperature. Even though 1/1000 of a drop of a 300 SUS *oil*** per hour will completely cover and lubricate the contacting surfaces of 15 to 50 mm bore bearings running at 3600 r.p.m.[3], this amount must continually be replenished in operation.

On the other hand laboratory studies using dye tracers in shielded bearing lubrication have shown substantial feeding of the *grease* proper to the bearing[4]. Also, examinations of sealed traction-motor armature bearings have indicated some migration of grease from end-plate reservoirs to the bearing in service. It is possible that the particular temperature-speed-vibration conditions in question affect the nature of the lubricating phase or phases fed to the bearing. Bearing and housing geometry are also factors in this situation, as is grease composition.

Higher Temperatures

There have been great increases in the life which lubricating greases now provide at elevated temperatures. This is the result of an intensive study of oxidation-resistant formulations. Such research has gone hand in hand with a fundamental study of lubricating grease structure by means of the electron microscope, X-ray diffraction equipment and optical microscopy. By means of modern knowledge the grease chemist can now scientifically arrange the lubricant structure so that improved feedability and long life are obtained.

Figure 1 shows the eight- to ten-fold increase in life at elevated temperatures which has been made available by improved greases during the past ten years[5]. Lives predicted by these laboratory rig results may be lowered for service conditions where ambient moisture, dirt and poor maintenance are factors; however, the laboratory results are comparative among themselves and so indicate the great improvement made.

Figure 2 shows the improvement made in maximum temperature of operation for petroleum greases in electric motors during the period 1947 to 1957[5]. It may be seen that the maximum temperature for one-year "bearing life" has risen from approximately 175°F. to 250°F. The "bearing life" shown in Figures 1 and 2 is governed essentially by grease life as these were not highly loaded tests.

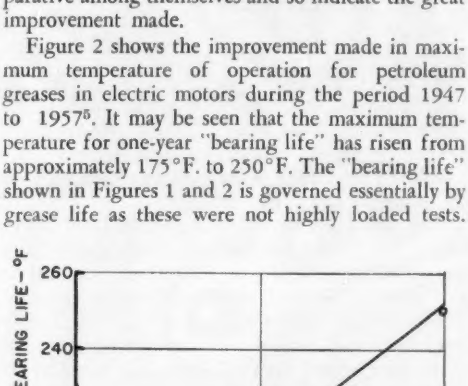


Figure 2 — Improvement in Maximum Temperature for Petroleum Greases in Electric Motors (reprinted from 1957 article "Recent Advances in Grease Lubrication of Ball-Bearings" by E. R. Booser[5]).

* Greases are both thixotropic and non-Newtonian. A non-Newtonian material decreases in viscosity with *increasing* rate of shear with this effect being immediately reversible with decreasing shear rate. On the other hand, thixotropic hardening is proportional to aging time of the undisturbed grease.

** Viscosity in Saybolt Universal Seconds at 100°F.

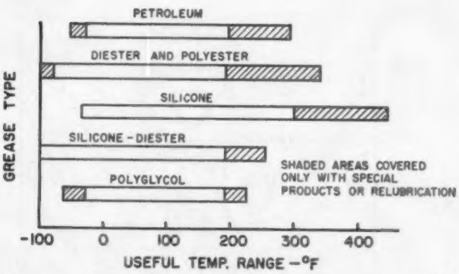


Figure 3 — Temperature Range for Ball-Bearing Greases (reprinted from 1957 article "Recent Advances in Grease Lubrication of Ball-Bearings" by E. R. Booser[5]).

Considering even higher temperatures, a cooperative program at 300°F. using silicone-insulated motor/generator sets has yielded much valuable information concerning grease lubrication of No. 310* ball bearings. A comprehensive study of the factors involved, made by government laboratory personnel, indicates among other things that the design of the end-bell (which houses the reservoir grease) is important if maximum utilization of the grease charge is to be secured[6].

Aircraft needs in particular have raised operating temperature requirements to the 450°F. and higher level. Testing at such extreme high temperatures is discussed later in this article.

Extended Temperature Range

Figure 3 illustrates the tremendous temperature range over which grease lubrication may now be achieved, as a function of general composition of the lubricant[5]. It may be seen that synthetic oily components are utilized to achieve the lowest temperatures of operation down to −100°F. and also the highest temperatures up to 450°F. At the high-temperature end, the usefulness of synthetic materials, such as high-molecular-weight esters and silicone fluids, has been extended and made more effective by the use of unusual thickening agents.

It has been found that soap-thickened greases do not have the maximum heat stability needed for the extreme high temperature range and, therefore, many unusual materials have been investigated and are being used for such very high temperatures. These include organic modifications of clays, certain very stable dye-stuffs and complex organic compounds such as the arylureas[5].

Higher Speeds

Important advances have also been made in anti-friction bearing lubrication at high speeds. In the past, a speed of 10,000 RPM was considered as high, however, RPM's of 50,000 to 150,000 are not unusual at this time in certain aircraft and machine tool applications[31, 7], mostly with oil mist lubrication.

Revolutions per minute is not the only measure of speed; one must consider the peripheral speed as being most important. A comparative measure of peripheral speed is the so-called DN Value (bearing bore in millimeters times speed in revolutions per minute) which gives a speed factor. For instance, a No. 204 bearing (20 mm bore) at 10,000 RPM has a speed factor or DN Value of 200,000.

Experiments carried out at a major petroleum research center over the last several years have resulted in the development of greases which allow substantial bearing life for speeds up to 50,000 RPM for a No. 204 bearing; a DN Value of one million. This work has yielded valuable information on the types of greases which are suitable for such speeds and also some information on the types of ball-bearing design which are most appropriate[8, 9].

Bearing life is usually defined as the time at which 10% of the bearings have failed through fatigue. For a given bearing, doubling the speed halves the expected life of the bearing. Doubling the load reduces the life to one-eighth.

It has been emphasized recently that centrifugal load is a very important factor in ball-bearing life at very high speeds. Certain authorities[1] have arranged in graphical form a number of derived relations[10] on the effect of speed on equivalent radial load. Such information is presented in Figures 4a, 4b and 4c, and it may be seen that bearings become very heavily loaded from this effect when speeds of 30,000 to 50,000 RPM are achieved. This helps to explain why such a "scatter" of results is obtained on evaluating lubricants at such speeds. The bearings are loaded to such an extent that fatigue life is materially reduced. This factor is in addition to failures from cage wear and breakage.

Better Rust Protection

Another achievement in grease formulation has been the development of effective rust inhibitors. This has been concurrent with the development of suitable test techniques as will be discussed later. The development of rust-inhibited formulations has been particularly valuable for greases based on water-repellent soaps, such as calcium and/or lithium. Greases made from such soaps are extremely resistant to water washout, however, unlike certain sodium soap greases[11], they require additional fortification for optimum rust protection.

The need for a high degree of protection against rust is extremely important in applications where

*A typical ball bearing number, such as 310, conveys considerable information. For instance, the last two digits give the bearing bore number, which is 1/5 of the bore size in millimeters when the bearing bore is 20 mm or larger, thus giving a bore size of 50 mm for this bearing. The digit 3 is related to the outside-diameter series which is also related to the load capacity of the bearing. However, it is necessary to consult tables to find the outside diameter and width of bearings of a certain series.

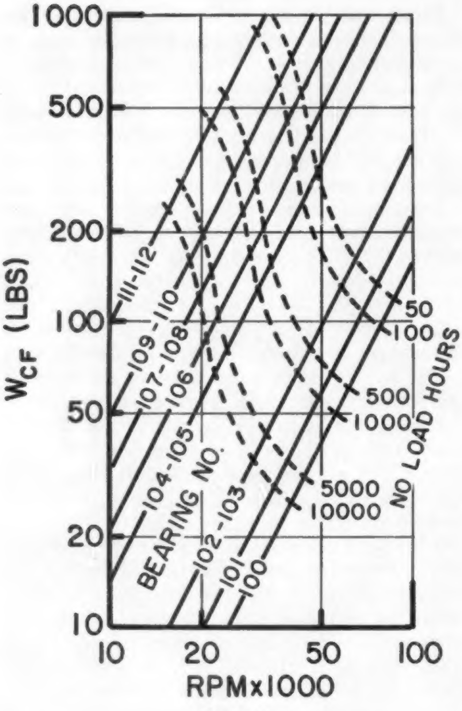


Figure 4A

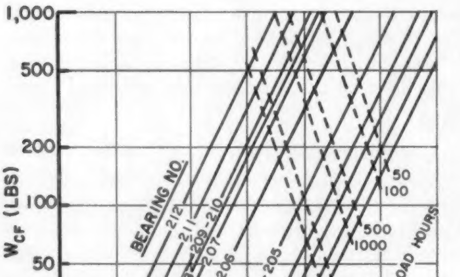


Figure 4B

Figure 4a, 4b, 4c — Equivalent centrifugal load Wcf, as a function of speed for (a) 100-series, (b) 200-series, and (c) 300-series ball bearings. Dashed lines indicate constant bearing life at zero external load. ("By permission from BEARING DESIGN AND APPLICATION by Wilcock and Booser. Copyright, 1957. McGraw-Hill Book Company, Inc.")

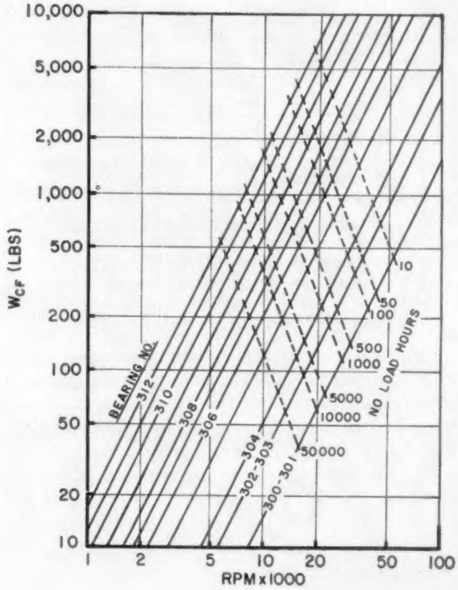


Figure 4C

rolling-element bearings must stand for long periods of time in the presence of humid atmosphere. This may occur under adverse conditions of storage of prepacked bearings such as at tropical military depots, and it also may occur in wheel bearings during standing of vehicles or railway cars for extended periods.

Beyond the Horizon

Grease research is always reaching out and/or being pushed into new frontiers by military and industrial break-throughs. Perhaps the newest frontier at the present time is that of resistance to nuclear radiation. Such lubricants are of pronounced interest to the military and are becoming more important in industrial applications as atomic power plants go into operation.

In general, greases first soften on irradiation because of disintegration of the soap structure, and finally harden to porous and brittle substances owing to polymerization (combining of small molecules to form larger ones) and cross linking of the oil molecules. Performance life in bearings is reduced concurrent with reduction in oxidation resistance.

Such locations as remote fuel-handling devices, control rod drives, coolant pump bearings and accessory units will need greases with greater resistance to radiation as times goes on [12]. Current information indicates that such greases will probably contain an oily component high in aromatic content and that non-soap thickeners will probably be used[13, 14].

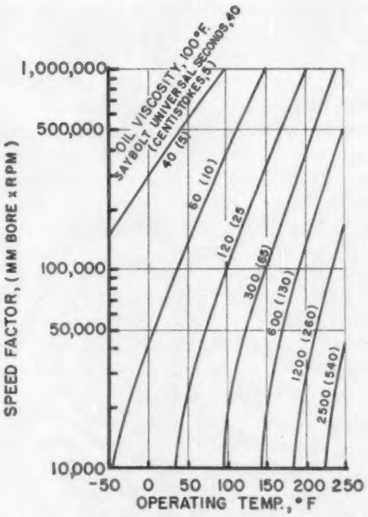


Figure 5 — Recommended oil viscosities for ball and roller bearings. ("By permission from BEARING DESIGN AND APPLICATION by Wilcock and Booser. Copyright, 1957. McGraw-Hill Book Company, Inc.")

OIL LUBRICATION

The advantages of grease were pointed out earlier. However, oil is more effective in carrying away heat. Oil also affords more positive feeding to loaded contact areas of bearings and furnishes a flushing action for any dirt, water or wear products. A correlary to this last fact is that the oil may be purified in a circulating system so that long periods of operation under adverse conditions may be achieved, whereas this would not be possible with grease in such applications as roller bearings on the dryer ends of paper mill machines[15].

In many cases the requirements of other machine elements dictate the selection of type and viscosity of the lubricating oil used in the anti-friction bearings. It is stated by current authorities[1] that "if other factors do not dictate the oil properties, the viscosity of the oil for a ball or roller bearing should be just high enough to insure satisfactory lubricating performance for the rolling elements and the separator. Higher viscosities will result in higher operating temperatures and higher power dissipation than necessary". These authors have made a composite of lubricant recommendations available in the recent literature with those of the bearing suppliers as a guide for selecting the proper oil viscosity for ball or roller bearings. This graph is reprinted as Figure 5.

Compositionwise, *mineral* oils are still quite adequate under most conditions for lubrication of anti-friction bearings and are used extensively. However, for extreme conditions of low temperature and high temperature, or wide temperature range, *synthetic* oils are used. This is particularly true in modern turbine engines where the bearings are lubricated with synthetic oil of the so-called diester type. These bearings must be lubricated to permit military engine starting at —65°F and to give satisfactory operation at temperatures of the order of 400-500°F in some applications. In the future, the upper bearing operating temperature level is expected to increase. Accordingly, lubricant suppliers are continuing to devote considerable effort to the development of lubricants which will be superior to current fluids at elevated temperatures. A prime problem in this regard is the formation of bearing deposits resulting from this exposure of oil to high temperatures in the presence of air. The high temperature laboratory bearing tests now being developed should be of material assistance to those developing superior high temperature oils. Whether the superior high temperature products will possess the excellent low temperature properties of present oils is debatable; it appears that if improvement is to be obtained at high temperatures some sacrifice in low temperature performance may be required. It is of interest to note that there is some evidence that certain mineral oils may be superior to synthetics at elevated temperatures. The surface deterioration of bearing working areas at elevated temperatures is also under investigation. The relative performance of different oils with respect to this factor is not at all well defined at present. Depending upon test conditions, synthetic oils may or may not be found superior to mineral oils[16, 17, 32]. Clearly, additional work will be required in this field to better understand the various ramifications involved.

ADVANCES IN TEST TECHNIQUES

1. Cooperative Developments

The Coordinating Research Council has continued with its development of the 10,000 RPM spindle test for greases which uses a No. 204 test bearing. This test is now called for in military specifications at temperatures up to 450°F. (MIL-G-25013A) and precision is quite good even at the highest temperature. At 450°F. and above it is necessary to use special high alloy tool steel bearings to avoid dimensional changes. The CRC is now continuing its work at the 600-700°F. level in view of continuing demands from the Military that evaluation methods at such temperatures be made available.

In the field of fretting corrosion, considerable advance has been made in developing a suitable instrument for determining the effect of lubricating greases in minimizing this form of wear. The August, 1955 issue of this publication was devoted to this subject and contains on page 93 a photograph of the Sikorsky apparatus for oscillating radially loaded bearings, such that the effect of lubricants on their fretting corrosion tendencies may be determined. A

Figure 6 — ASTM Wheel Bearing Tester. Apparatus, which employs a 1942 Ford front wheel hub is driven at a speed equivalent to 40 miles per hour. Test bearing assembly is enclosed in insulated hood so tests can be made at elevated temperatures. Special ring is used in place of regular grease retainer, to catch any grease which leaks from the inner end of the hub.

very recent publication brings this work up to date[19].

A much needed development has been the final publication of a technique for evaluating the rust-resistant properties of lubricating greases. This work is the culmination of a number of years of cooperative investigation along these lines and has resulted in a simple bearing protection test technique. In this test a small tapered roller bearing is packed with the grease to be investigated and placed in a jar in the presence of a small amount of free water. The jar is then closed and the assembly kept for fourteen days at 77°F. At the end of this time the bearing is cleaned and examined for rusting. This test is described in Military Specification MIL-G-3278A, "Grease, Aircraft and Instrument." Correlation was obtained between the Bearing Protection Test and two full-scale field tests in different parts of the U.S.A. on fleets of aircraft in which the test greases were placed in the wheel bearings of the aircraft[20].

The ASTM* has standardized the automotive wheel bearing performance test originally developed by the CRC and this test has been in military specifications such as MIL-G-2108 for some years. It measures the leakage tendencies of greases; however, being a short-time (6-hour) test it does not evaluate the lubricating abilities of the grease. Figure 6 shows a photograph of this test unit. The necessity of full-scale field tests as a proof of automotive wheel bearing service is well proven by a comprehensive study reported in 1954[20].

*"American Society for Testing Materials"

Compatibility of lubricating greases has received considerable attention during recent years. It has been found that mixtures of certain greases may show softening and high leakage under certain conditions from physical structure changes. It is also possible for chemical interactions to take place. For instance, when a grease containing high free alkali content is mixed with an ester-type synthetic grease, saponification of the ester may occur resulting in an unsatisfactory lubricating mixture. While various short-time tests have been proposed as suitable for testing compatibility of grease mixtures, the only safe way is to run such blends in antifriction bearing machines under the same type of conditions which will be experienced in the field. Such tests must be carried out for considerable periods of time, as short-time tests may not predict latent incompatibility.

The ASTM has considered this question and Technical Committee G of ASTM Committee D-2 has issued the following statement:

"STATEMENT ON COMPATIBILITY TESTING OF LUBRICATING GREASES

Compatibility of lubricating greases implies compatibility in service. Owing to the complexity of service problems, there is no single test which can be used to satisfactorily evaluate compatibility in general.

Present ASTM Methods, such as those for Dropping Point, Worked Penetration, and Leakage Tendencies of Automotive Wheel Bearing Greases, may be used to test individual greases. Results obtained on such mixtures, as compared to results on the unmixed component greases, are indicative of the compatibility of the mixtures. However, simple bench tests such as these must be correlated with a particular set of service conditions before a valid estimate of compatibility in such service can be achieved."

As an example of a rigorous mechanical test for compatibility, the Association of American Railroads (AAR) has issued Specification No. M-917-56 where greases submitted for qualification must be run with mixtures of approved products in an eight-week simulated service test. A four-axle tester with two roller bearing assemblies (6x11 boxes) per shaft is used. The shafts turn at 640 RPM which is equivalent to 63 MPH for a 33-inch freight wheel. The tests run for eight weeks with periodic shut-downs. Fan blades on the shafts provide cooling to simulate the windage of road operation. A 600-lb. vertical load is imposed on each bearing.

The development of suitable greases for railroad axlebox roller bearings has been an outstanding development of the past decade. Reduced maintenance has meant lower labor costs. Tests have shown that freight car grease-lubricated bearings can

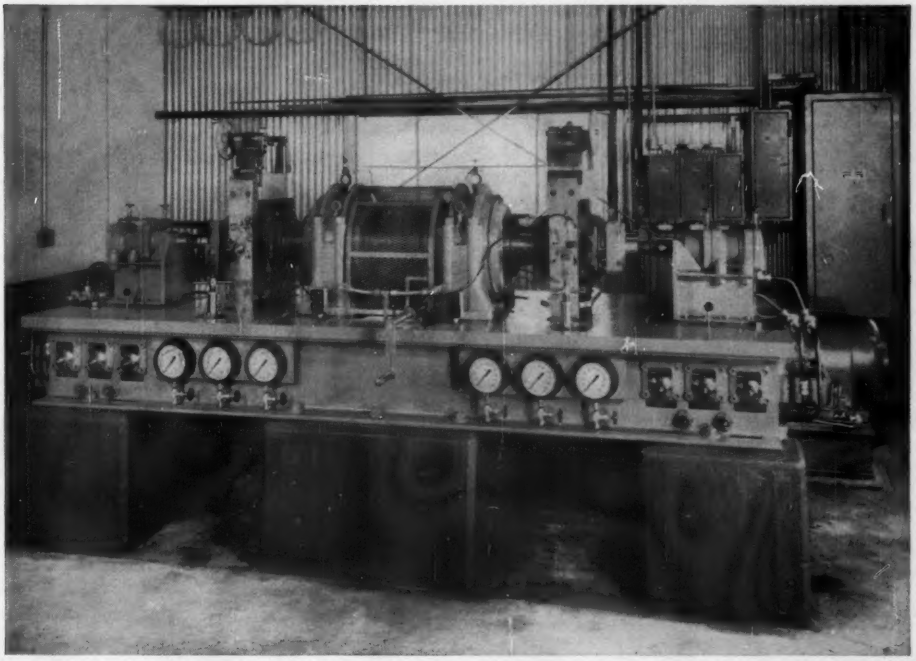


Figure 7 — Seven-ton, full-scale Railroad Journal Bearing Test Machine developed by a prominent petroleum research laboratory showing two actual $5\frac{1}{2}$ x 10 inch journal boxes assembled with vertical loading yokes and axial load cylinders.

operate for at least three years without relubrication[21]. This whole picture is well summarized in a very recent comprehensive paper by a representative of a large roller bearing manufacturer[21].

Individual Developments

Some years ago a large petroleum research laboratory developed a highly versatile Railroad Journal Bearing Tester which has been in constant and useful operation since its installation. Figure 7 shows this apparatus which tests two full-scale roller bearing assemblies at one time. Heavy vertical and thrust loads are imposed, together with superimposed shock loading, all activated hydraulically. Lateral play is provided for as in actual service, and friction measurements may be made at the start of and during a run. This machine has aroused much interest in domestic and European railroad circles and similar equipment has been constructed by the AAR.

A large manufacturer of electrical equipment has been running for a number of years an interchangeable-cartridge high-temperature ball-bearing grease tester which was described in detail at the ASTM Symposium on Functional Tests for Ball Bearing Greases in 1948[2]. Tests are run up to 10,000 hours at 212°F. ambient temperature. This test utilizes a No. 306 bearing with a 160 pound radial load and 40 to 50 pounds thrust load and runs at 3600 RPM. It is considered to give a good simulation of electric motor bearing operation, as far as grease life goes, under laboratory conditions. Figure 8 shows a sectional view of this bearing tester.

A large manufacturer of rolling-element bearings has developed valuable correlations from temperature-load-speed studies on ball bearing life on test rigs of their own design[22].

A recent publication describes the scope of evalu-

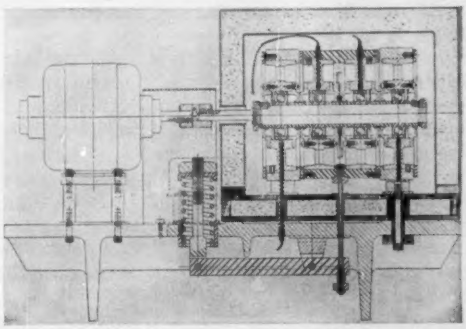


Figure 8 — sectional view of G.E. Four-Bearing Interchangeable Cartridge High-Temperature Ball Bearing Grease Tester. (reprinted from article "Grease—An Oil Storehouse for Bearings" by D. F. Wilcock and M. Anderson, *Symposium on Functional Tests for Ball Bearing Greases, STP-84, ASTM*[2]).

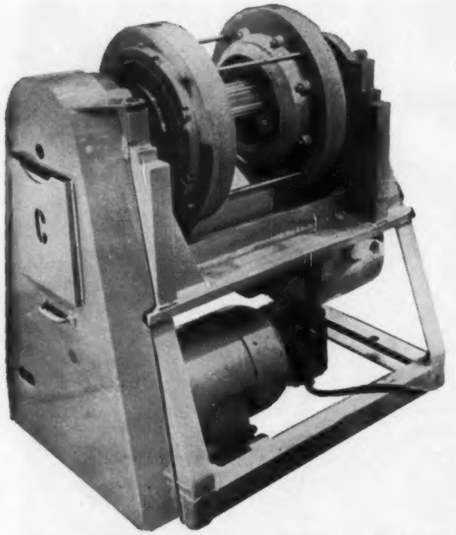


Figure 9 — EMD Traction Motor Armature Bearing Tester. (utilizes 130 mm bore test bearings; representative test conditions are 500-hour duration, 2500 RPM, room ambient temperature and 740-lbs. load per bearing. Superimposed vibration is 800 CPM.)

ation and testing imposed by a major petroleum company on the various greases developed[23].

For testing the performance of roller bearing greases under heavy loads the SKF Roller Bearing Test Machine continues to be of particular value [24]. This apparatus tests 5½ inch OD cylindrical roller bearings under 4240 pounds per bearing radial load and a usual speed of 2100 RPM. By means of cartridge heaters under the test bearings it is possible to run at elevated bearing temperatures. The value of this machine in studying the performance of different grease types, as well as further studies with the Vibrating Wheel Bearing Tester and the Torque-Breakdown Machine are described in a paper published in 1955[24].

In Europe the SKF R2F Test is widely used, in which spherical roller bearings of 60 mm bore are used as the test bearings. Various conditions of speed and temperature are imposed on the lubricating greases to be evaluated. This type of bearing, which has barrel rollers and a curved outer raceway, possesses high load-carrying capacity. Because of the greater degree of contact (osculation) between roller and outer raceway, this design constitutes a severe test for lubricating greases particularly at increased speeds.

There has been considerable test development in the field of traction-motor-armature roller bearings for railroad use. The use of sealed bearings with the initial bearing lubricant charge lasting between overhauls with no relubrication is now an accepted procedure and widely used. Periods of 300,000 to 500,000 miles between relubrications of such bearings are now being achieved in railroad service. Whereas formerly flange-riding-roller bearings were used, the trend is now to roller-riding-cage roller bearings. Figure 9 shows a picture of the EMD design of traction motor armature bearing tester.

Miniaturization, radiated heat from jet engines and aerodynamic heating at supersonic speeds are three of the important factors in the military picture which have pushed grease operating temperature requirements to 450°F. and higher. As early as 1949 the USAF* asked a subpanel of the Coordinating Research Council to screen existing aircraft control bearings and lubricants for life at 500°F. This work led to the development of an oscillating control-bearing tester in which tests were run up to 550°F. under conditions of load, degree of oscillation, and speed representative of a jet fighter at that time. When conventional KP-16A bearings were fitted with heat-resistant seals it was possible to get 20 to 80 hours life at 550°F. with a non-soap, solid-thickened/silicone fluid grease and up to 248 hours at 500°F. with this grease[25]. Figure 10 shows a picture of this apparatus.

The field of extreme low temperatures has not been neglected. In one detailed study, an intensive program[26] has been carried out to note the degree of correlation of torques measured by simple laboratory bench rigs with the results obtained on full-scale aircraft accessories at temperatures down to −100°F. Figure 11 shows a combined view of the various pieces of accessory units which were evaluated in this study and Figure 12 pictures a retractable landing light with the auxiliary equipment necessary to run these tests.

The conclusions from this work were that there were more variations between units of supposedly identical construction than there were between greases meeting the same military specification which in this case was MIL-G-3278A, "Grease Aircraft and Instrument." A certain accessory which was overpowered, operated at —100°F. even with a grease which was designed only for −40°F. operation (the criterion being one revolution of a No. 204 bearing at 5 seconds or less at 2,000 gram-centimeters torque). A unit of another accessory was under-powered and would not operate at −65°F. even with the grease designed to work at −100°F. as judged by the 2,000 gram-centimeter criterion. Therefore, apparently large numerical differences detected by a sensitive laboratory bench test may have no significance for actual mechanisms where a number of bearings, gear trains, etc. are involved, and where the power available to cause operation of the unit is a very important factor.

Lubricating grease performance in the presence

*United States Air Force

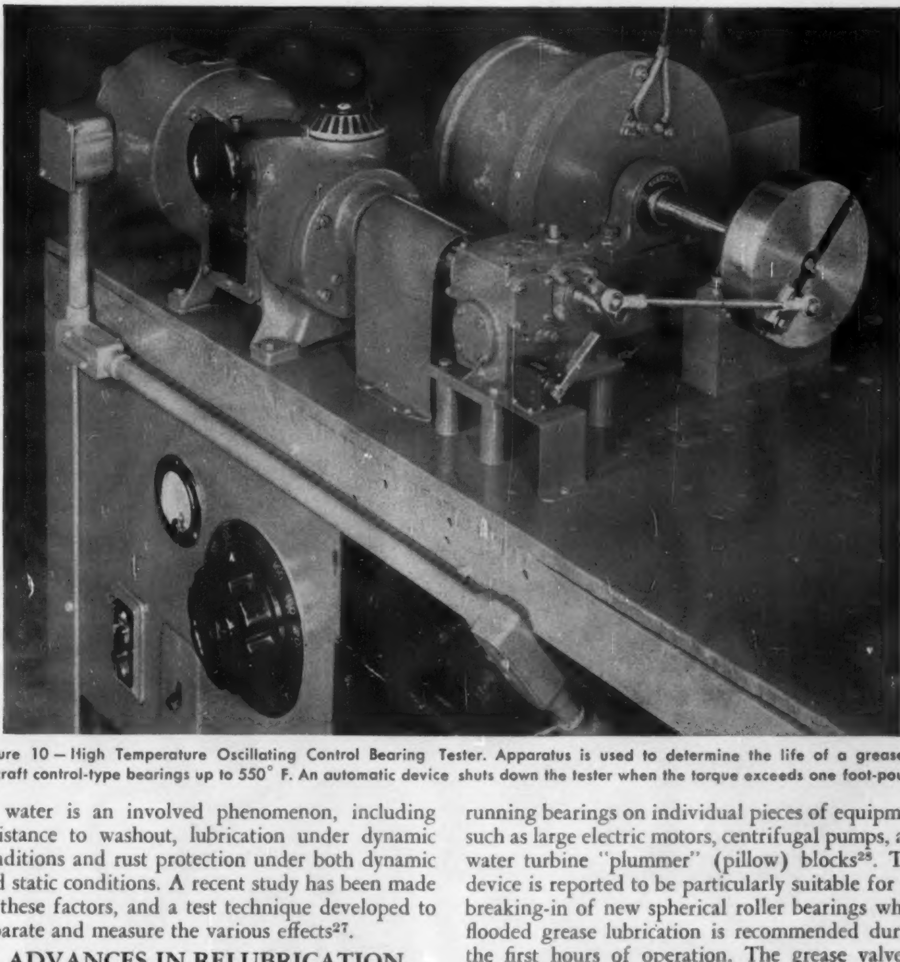


Figure 10 — High Temperature Oscillating Control Bearing Tester. Apparatus is used to determine the life of a grease in aircraft control-type bearings up to 550° F. An automatic device shuts down the tester when the torque exceeds one foot-pound.

of water is an involved phenomenon, including resistance to washout, lubrication under dynamic conditions and rust protection under both dynamic and static conditions. A recent study has been made of these factors, and a test technique developed to separate and measure the various effects[27].

ADVANCES IN RELUBRICATION

1. Greases

The foregoing sections have described the great advances in sealed and cartridge-type bearings, where one fill of grease lasts for the life of the part or to a major overhaul. Yet there are many applications such as in the metal industries and in industrial machine lubrication, where periodic relubrication — "the right amount of the right grease at the right time" — is a must.

The August 1957 issue of this publication gives a comprehensive and up-to-date picture of automatic grease dispensing equipment and pertinent information on flow properties of greases and so forms a valuable supplement to the issue at hand[30].

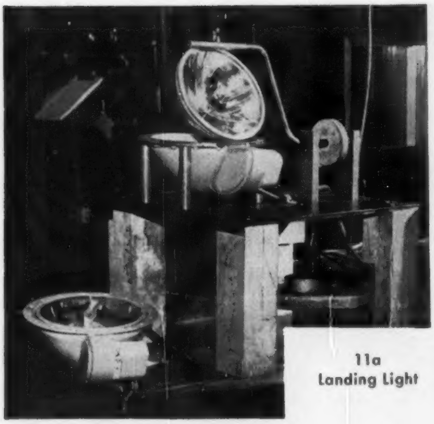
A further development which has taken firm hold in Europe during recent years is that of the so-called "grease valve" which expedites the relubrication of running bearings on individual pieces of equipment such as large electric motors, centrifugal pumps, and water turbine "plummer" (pillow) blocks[25]. This device is reported to be particularly suitable for the breaking-in of new spherical roller bearings where flooded grease lubrication is recommended during the first hours of operation. The grease valve is essentially a rotating slinger which directs grease purged through the bearing into a venting passageway so that excess grease will not pile up in the bearing and cause overheating, churning and grease breakdown.

Figure 13* shows a drawing of a grease valve arranged for electric motor bearing relubrication. Figure 14* shows how the valve causes a rapid drop in temperature following successive relubrication periods as compared to operation without the valve.

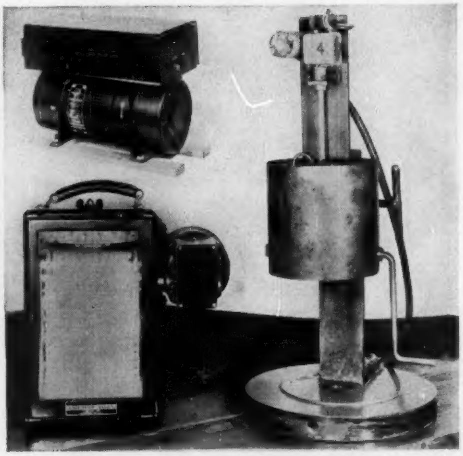
2. Oils

A companion article to the August 1957 issue is that of November, 1957 on Central Air-Borne Lubrication Systems which describes the unconventional but very important systems for application of lubricating fluids as a spray or more finely atomized mist.

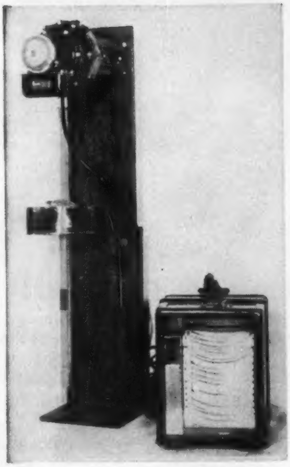
*See page 112



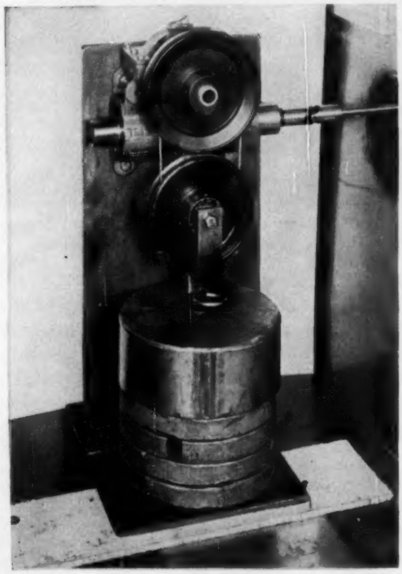
11a
Landing Light



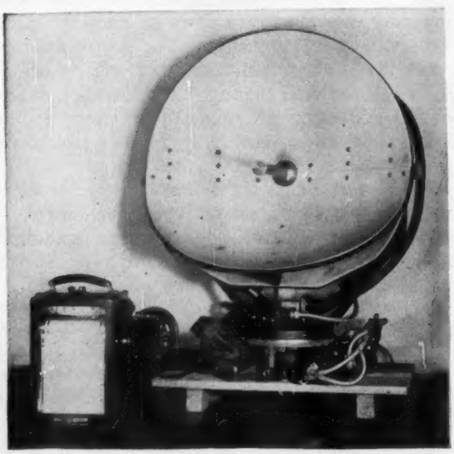
11b
Actuator Assembly
Shutter-vane Intercooler with 400-cycle Inverter



11c
Actuator Assembly
Bomb-Bay Door



11d
Gear Assembly, Elevator Flap Tab



11e — Antenna Assembly

Figure 11 — Composite view of five pieces of aircraft accessory equipment tested at temperatures down to —100°F.

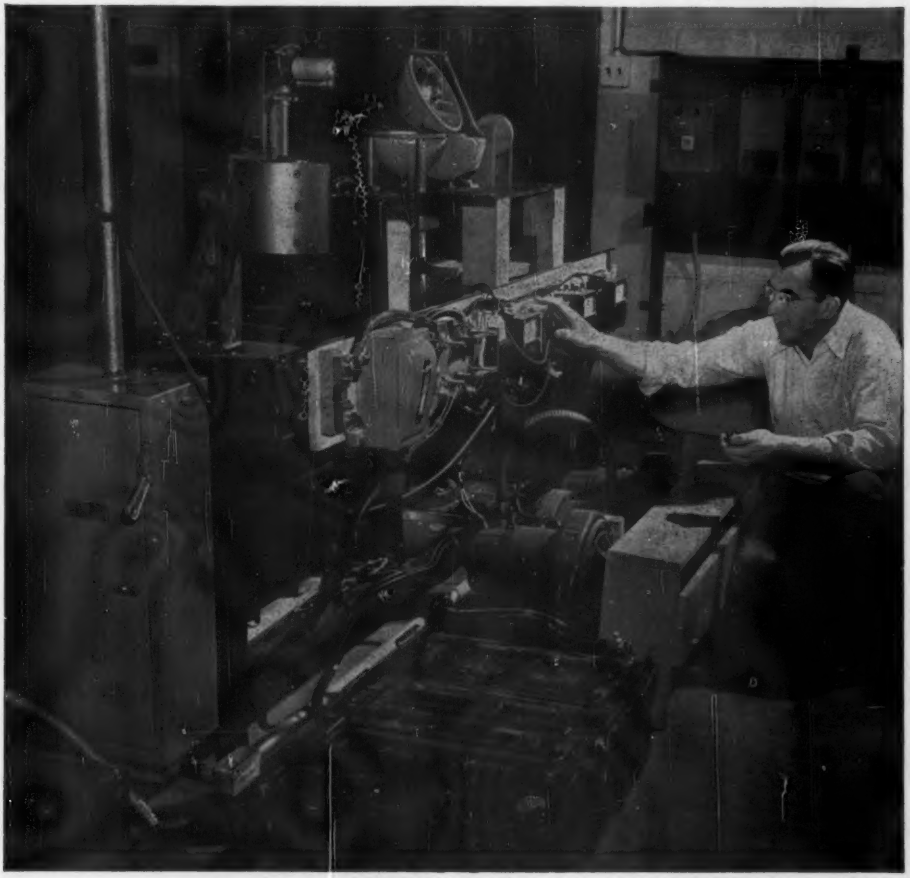


Figure 12 — Aircraft retractable landing light assembly with associated electrical equipment necessary to operate the landing light at temperatures down to —100°F.

SUMMARY

It is difficult to overemphasize the importance of rolling-element bearings to our economy. The petroleum industry is proud to have been a companion in the development of present bearing/lubricant systems upon which so much of industry and our defense effort depend.

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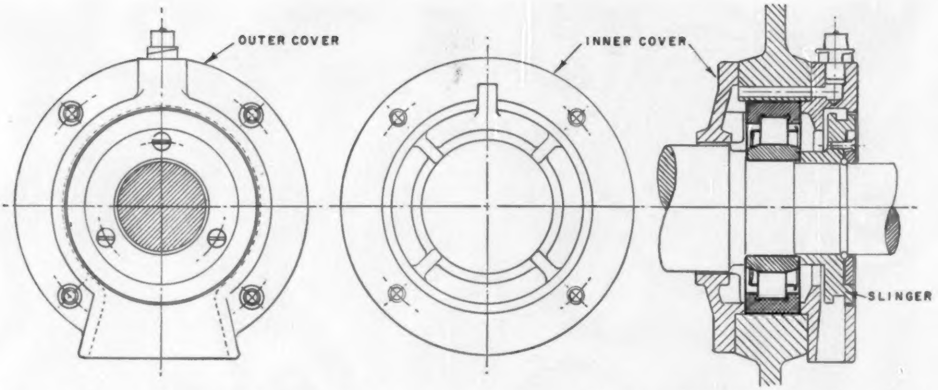

Figure 13 — Grease valve in electric motor with end frames (reprinted from SKF brochure TSP 5243[28]). Note design of bearing housing covers which incorporate radial webs to lead new grease into the bearing.

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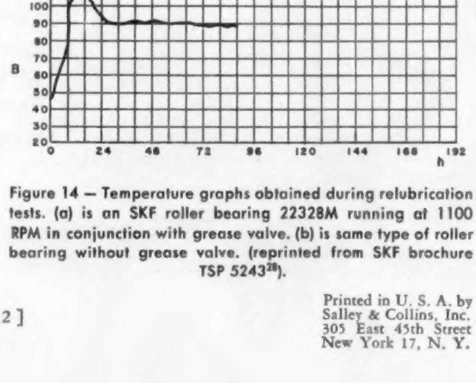

Figure 14 — Temperature graphs obtained during relubrication tests. (a) is an SKF roller bearing 22328M running at 1100 RPM in conjunction with grease valve. (b) is same type of roller bearing without grease valve. (reprinted from SKF brochure TSP 5243[28]).

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SYSTEMATIC ROLLING-BEARING TROUBLE-SHOOTING CHART

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Cause	*Remedy*
1. OVERHEATED BEARING:	
a. Inadequate lubrication	Change to proper grease or oil. Adjust oil-cup height, maintain oil level at center of lowest ball or roller. Fill grease housing ½ to ⅔ full. Clean oil holes, filters, and vents. Use a fresh lubricant.
b. Excessive lubricant churning	Use lower-viscosity oil, lower oil level to center of bottom ball or roller, fill grease housing only half full, use oil mist
c. Inadequate internal clearance	Use bearing of greater looseness, allow for differential thermal expansion, reduce interference of shaft and housing fits, correct any housing out of roundness or warping
d. High seal friction	Stretch felt or use reduced spring tension with leather or composition seals, lubricate seals, switch from rubbing seal to low-clearance shield
e. Excessive preloading	Use gaskets or shims to relieve axial preload with opposed pair or with two held bearings on a shaft subjected to thermal expansion. Change design to use only one held bearing
f. Spinning outer ring	Use closer housing fit, use steel insert in soft aluminum housing, use garter spring or rubber holding ring
g. Misalignment	Correct alignment by shimming pillow blocks, housings, or machines to get shafts and bearings in line. Check for misalignment of bearing seats and shaft and housing shoulders
2. NOISY BEARING, VIBRATION:	
a. Wrong type of grease or oil	Check recommendations of lubricant supplier. A higher-viscosity oil or a better-feeding grease with a higher-viscosity oil may help
b. Insufficient lubrication	(See 1*a*)
c. Defective bearing	Check for brinelling, fatigue, wear, groove wobble, poor cage. Replace bearing
d. Dirt	Clean bearing housing, replace worn seals, improve seal arrangement, eliminate source of dirt
e. Corrosion	Improve sealing to keep out corrosive elements, use corrosion-resisting lubricant
f. Too great internal clearance	Change to bearing with smaller clearance
g. Unbalance	Balance rotor
h. Misalignment	Align (see 1*g*)
i. Too loose shaft or housing fit	Build up shaft or bore with chrome plate or metallize and regrind
j. Improper mounting	Correct dirty or off-square shaft and housing shoulders and seats. Avoid brinelling caused by pounding on bearing
k. False brinelling	Use vibration mounts for machine to isolate from platform during idle periods
l. Seal rub	Check for metal bearing seal or shield rubbing on shaft, shoulder, or housing
3. LOSS OF LUBRICANT:	
a. Oil leakage through seal	Adjust oil level to center of lowest ball or roller, replace seal, use double-seal arrangement with drain between, eliminate any unfavorable air flow by proper baffles and balancing channels
b. Leakage at housing split	Use thin layer of gasket cement, replace housing
c. Grease leakage	Pack housing only ½ to ⅔ full, use channeling-, type grease, eliminate any pressure causing air flow through bearing, keep solvents or water from entering and softening grease, use new or improved seals
d. Dry, caked residue	Use silicone or other high-temperature grease, use oxidation-inhibited or synthetic oil or grease, cool oil in external cooler, cool bearing housing, increase oil flow to promote cooling.
4. LOOSE BEARING:	
a. Shaft diameter too small	Turn down shaft, chrome-plate or metallize and regrind to give proper fit. Retighten adapter to get firm grip on shaft
b. Housing bore too large	Build up bore with chrome plate or metallize and regrind, bore out housing and press in sleeve to give proper bearing fit (a slip fit on OD is usually desirable to allow for differential axial thermal expansion of a shaft between two bearings)
5. HARD TURNING OF SHAFT:	
a. Excessive bearing preload	Use less interference fit on shaft or in housing, select bearing with greater internal clearance where heat conduction expands shaft and inner bearing ring. Relieve axial preloading by housing shims with either two opposed bearings or two "held" bearings on one shaft
b. Heavy seal rub	Stretch felt seals to reduce their friction, use reduced spring tension on leather or composition seals, scrape out ID of rubber seals, use a shield with clearance on the shaft
c. Dirt	Clean housing and use fresh lubricant (see 2*d*)
d. Corrosion	(See 2*e*)
e. Lack of lubrication	Raise oil level to center of lowest ball or roller, grease pack housing half full. Eliminate possibility for loss of lubricant by leakage
f. Incorrect lubricant	Use softer grease or thinner oil
g. Bearing pinching or cocking	Scrape housing bore to relieve pinching. Replace or remachine warped housings, check bearing seats as source of cocking